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# Aerodynamics and Heat Transfer Investigations on a High Reynolds Number Turbine Cascade

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# AERODYNAMICS AND HEAT TRANSFER INVESTIGATIONS ON A HIGH REYNOLDS NUMBER TURBINE CASCADE

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## ABSTRACT

In this report the results of aerodynamic and heat transfer experimental investigations performed in a high Reynolds number turbine cascade test facility are analyzed. The experimental facility simulates the high Reynolds number flow conditions similar to those encountered in the space shuttle main engine. In order to determine the influence of Reynolds number on aerodynamic and thermal behavior of the blades, heat transfer coefficients were measured at various Reynolds numbers using liquid crystal temperature measurement technique. Potential flow calculation methods were used to predict the cascade pressure distributions. Boundary layer and heat transfer calculation methods were used with these pressure distributions to verify the experimental results.

## NOMENCLATURE

| | |
|---|---|
| c | blade chord |
| h | heat transfer coefficient |
| p | pressure |
| P/Ptot | static to total pressure ratio |
| $r_{LE}$ | blade leading edge radius |
| $r_{TE}$ | blade trailing edge radius |
| Re | Reynolds number based on blade chord |
| S/Smax | ratio of blade surface distance measured from the leading edge to trailing edge for each surface |
| S/Stot | ratio of blade surface distance measured from the suction surface trailing edge to the pressure surface trailing edge around the blade |
| s | cascade spacing or pitch |
| T | total temperature |
| Tu | turbulence intensity |
| V | flow velocity |
| w | blade span or width |
| $\alpha$ | flow angle measured from the horizontal |
| $\delta$ | deviation angle from the flow angle |
| $\delta^*$ | boundary layer displacement thickness |
| $\Theta$ | boundary layer momentum thickness |

Subscripts

| | |
|---|---|
| s | static conditions |
| t | total conditions |
| 1 | inlet flow station |
| 2 | exit flow station |

## INTRODUCTION

Gas turbines used in the space shuttle main engine (SSME) turbopumps have been failing prematurely. The turbine blading develops cracks that require the turbines to be replaced more frequently than desired. The turbine cracking appears to be caused by extreme temperature fluctuations. These fluctuations are most extreme during engine start up and shut down (Abdul-Aziz et el., 1989). The nature of these temperature fluctuations and the resulting heat transfer is not fully understood.

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One contributing factor to the heat transfer problem encountered in the SSME is the external flow through the turbine blading. This flow is not typical of the flow found in aircraft gas turbines. The working fluid is hydrogen rich stream, and the pressure is very high. The high pressure causes the turbine blade Reynolds number and correspondingly the heat transfer coefficients to be an order of magnitude greater than those found in aircraft gas turbines.

An experiment was designed to provide basic heat transfer data at these higher Reynolds numbers. The experiment is intended to give turbine designers a better understanding of the heat transfer resulting from these flows. This knowledge should help designers in finding a solution to the SSME turbine failure problem, and provide insight into improving the design of future rocket turbopumps.

The high Reynolds number experiment has produced preliminary data. This data was analyzed using conventional inviscid flow solver and boundary layer techniques. The analysis was performed to verify the experimental design and to show the applicability of conventional heat transfer methods. Findings from the analysis will be used to improve the experiment before the final data is taken. The results from the experiment, the analysis, and recommendations to improve the experiments are reported in this paper.

## NECESSITY FOR EXPERIMENTAL RESEARCH

Quantitative prediction of the heat transfer behavior of turbine blades requires calculation methods that include governing parameters describing the physical conditions. For the design of turbomachinery blades, usually combined inviscid and viscous flow calculation procedures are utilized. Methods by McFarland (1985) and Katsanis (1977) can be used for calculation of the inviscid flow field. For the calculation of the viscous layer, integral or differential boundary layer calculation methods are used that in general contain correlations derived from flat plate experiments disregarding the free stream turbulence situation. To incorporate the turbulent characteristic into the boundary layer calculation procedure, Patankar and Spalding (1970) developed a differential method capable of solving the simplified time averaged Reynolds differential equation system consisting of conservation laws of continuity, momentum and energy. This method was improved by Spalding (1973), (1977) and tailored to the turbomachinery design needs by Spalding resulting in an advanced version called BTFT (Runchal, 1974). Crawford and Kays (1976) introduced a modified version of Spalding's program called STAN5. This code extended by Gaugler (1981) is capable of qualitatively predicting the heat transfer coefficient. Since the existing transition models incorporated in the code do not adequately describe the transition phenomenon under turbomachinery condition, a quantitative prediction remains questionable. Test studies reported by Gaugler (1985) show that, without the exact knowledge of the transition start point and length, with STAN5, a number of iterations must be performed until a reasonable agreement between computed and measured results is obtained. The results presented in this report indicate that the existing boundary layer calculation methods in integral or differential form cannot be considered as a reliable a priori predictive design tool unless the basic mechanisms mentioned above are completely described and implemented into the calculation procedure.

## EXPERIMENT

Since current heat transfer calculations do not predict heat transfer adequately and the SSME turbine Reynolds numbers are in a range where little heat transfer data exists, an experimental facility was designed and built to provide data at high Reynolds numbers. The high Reynolds numbers were achieved by building a large scale test blade. The blade size was chosen to get the largest Reynolds number possible using an existing one vane wind tunnel. Details of the experiment are given in a report by Yeh et el. (1990). A schematic of the test section is given in Fig. 1. Details of the test blade geometry and tested flow conditions are given in Tables 1 and 2 respectively.

The large scale of the experiment proved to be a benefit in instrumenting the test vane. Two blades were constructed. One was for aerodynamic data, and the other was for heat transfer data. The large size of the blade allowed inclusion of 52 surface pressure taps on the aerodynamic blade. This large number of surface pressure measurements provided detailed information about the blade loading and surface flow. For the heat transfer blade the large scale facilitated the use of liquid crystals for heat transfer measurements. The liquid crystal technique has been shown by Hippensteele and Russell (1988) to provide high resolution heat transfer data.

### Experimental Measurements

Both aerodynamic and heat transfer data were measured for three different Reynolds numbers ($Re = 1.33\times10^6$, $3.65\times10^6$, and $6.36\times10^6$). The upstream flow conditions were measured at a single mid-channel location. Total and static pressure, total temperature, and free stream turbulence were measured. The free stream turbulence for all three Reynolds numbers was 2.1 percent ($Tu = 0.021$). This is the natural turbulence level of the tunnel. The upstream flow conditions are given in Table 2. Experimental surface pressures were measured. Surface pressure data for two of the Reynolds numbers are shown in Figs. 3 and 4. The heat transfer data was taken at several locations along the blade surface. The liquid crystal technique allows data to be taken selectively along the blade surface. In this experiment the heat transfer data was concentrated in regions of rapidly changing heat transfer coefficient. Such regions usually indicate transition to turbulent flow. The heat transfer data was reported in terms of heat transfer coefficient, h. This data can be seen in Figs. 5 through 9. During the highest Reynolds number test, the heat transfer blade failed. The limited data taken at this Reynolds number should be considered less accurate when compared to the other data.

### Future Experimental Work

The experimental data taken to this point is considered preliminary. The test were carried out to nvestigate the experimental design and technique. Further testing is planned in the facility after refinements to the experiment have been made. The new tests will repeat the flow condition reported here in order to gain more confidence in the experiment. In addition the effect of increasing the free stream turbulence will also be investigated.

| Suction Surface x(m) | Suction Surface y(m) | Pressure Surface x(m) | Pressure Surface y(m) |
|---|---|---|---|
| 0.000000 | 0.000000 | 0.000000 | 0.000000 |
| 0.037727 | 0.027470 | 0.043945 | -0.026293 |
| 0.053849 | 0.027435 | 0.067185 | -0.037826 |
| 0.102601 | 0.007260 | 0.089764 | -0.050201 |
| 0.140494 | -0.026737 | 0.132255 | -0.078345 |
| 0.172551 | -0.068164 | 0.000000 | 0.000000 |
| 0.000000 | 0.000000 | | |

| | |
|---|---|
| Span: | $w$ = 0.215.9m |
| Chord: | $c$ = 0.28167m |
| Spacing: | $s$ = 0.16337m |
| Camber Angle: | $\alpha_2$ = -61.7° |
| Leading edge radius: | $r_{LE}$ = 0.02172m |
| Trailing edge radius: | $r_{TE}$ = 0.00381m |

Table 1: Geometric specifications of the single turbine blade heat transfer test facility

| Re | $\dot{m}$(kg/s) | $T_1$(°K) | $V_1$(m/s) | $p_s$(kPa) | $p_t$(kPa) |
|---|---|---|---|---|---|
| $1.33 \times 10^6$ | 3.76 | 294.9 | 16.98 | 264.34 | 264.62 |
| $3.65 \times 10^6$ | 10.31 | 290.3 | 45.87 | 263.84 | 266.45 |
| $6.36 \times 10^6$ | 17.59 | 285.8 | 75.66 | 268.91 | 276.17 |

Table 2: Inlet flow conditions used for the flow analysis

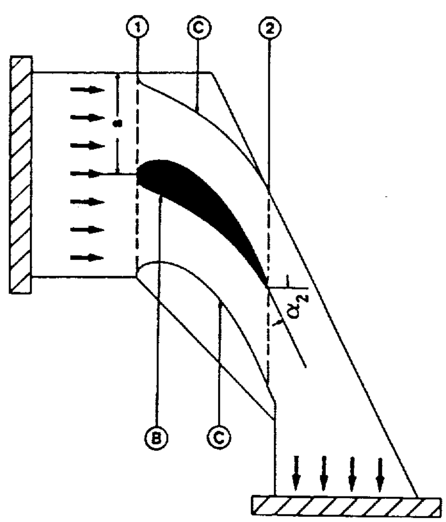


FIGURE 1. - TEST SECTION: B = SINGLE BLADE, C = CURVED CHANNEL WALLS S - SPACING, 1 = INLET STATION, 2 = EXIT STATION.

## ANALYSIS

An analysis of the experimental data was performed. The conventional inviscid-viscous flow calculation of combining a potential flow with boundary layer and heat transfer calculations was used. The primary purposes in making this analysis was to verify the experimental technique and identify improvements that could be made to the experiment. A secondary purpose was to investigate how well a conventional inviscid-viscous calculation method analyzes high Reynolds number turbine flows.

### Potential Flow Calculation

For the calculation of pressure distribution, inviscid flow calculation methods such as panel method by McFarland (1985) and finite difference calculation procedure by Katsanis (1977) have proved to generate reasonable results. Both methods use the equation of continuity for subsonic flow calculations. McFarland introduces a velocity potential and solves the resulting approximate compressible flow equations using an extended singularity method with variable strengths. As a solution of the these equations, the velocity distribution is obtained by treating the integral equation numerically and relating it to the pressure by an isentropic relation.

Katsanis uses the stream function method developed by Vavra (1968) and applies a finite difference method directly to find the solution to the partial differential equation for stream functions. Since the physical background for both methods are the same, similar results are anticipated. For the analysis presented, the singularity method was used for the calculation of velocity distribution necessary for heat transfer calculations.

### Results: Pressure Distribution

Before presenting the results, the specific configuration of the facility and its effects on the flow kinematics at the inlet and exit of the test section are discussed. The test section, shown in Fig. 1, has been designed to simulate a cascade flow, which is by definition a periodic flow. However the kinematic conditions at the inlet and exit differs from those in a cascade. Unlike the cascade flow, the flow in this test section experiences a sudden contraction at station 1 due to the existence of leading edges of the two adjacent side walls. By impinging on these leading edges, the flow generates corner vortices producing entropy that leads to higher total pressure losses. This effect has in connection with the heat transfer experimental program no significant consequences. At the exit station 2, the boundary layer flow on the convex channel wall encounters a sudden expansion that affects the exit flow angle. However, a correct exit flow angle is a fundamental requirement for an appropriate calculation of pressure distribution around a cascade profile. Under these circumstances, a reasonable agreement between a cascade flow calculation and the measurements was not expected.

Starting from the given trailing edge camber angle $\alpha_2 = -61.7°$ (Fig. 1), the inlet flow conditions are taken from Table 2 and the geometric specifications from Table 1. The pressure distributions were calculated for the inlet velocity $V = 75.66$ (m/s). As shown in Table 2, this velocity corresponds to the highest Reynolds number, $Re = 6.3\times10^6$. The calculation results are displayed in Fig. 2, where the relative pressure is plotted against the relative surface distance. As shown in Fig. 2, the calculated pressure distribution on the pressure surface agrees well with the experimental results. However, on the suction surface there is

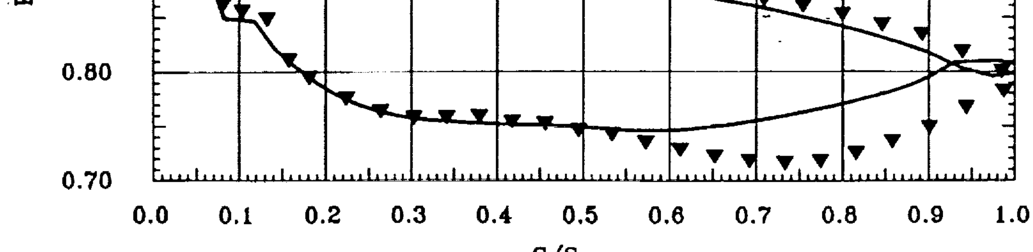


FIGURE 2. - MEASURED (▼) AND CALCULATED RELATIVE PRESSURE AS A FUNCTION OF RELATIVE SURFACE DISTANCE FOR THE EXIT CAMBER ANGLE $\alpha_2 = -61.7^0$ AND THE REYNOLDS NUMBER $Re = 6.3\times10^6$.

a significant discrepancy between experiment and calculation. This discrepancy is particularly pronounced in the second half of the suction surface and indicates that this portion is subjected to higher flow deflection than is established by the given blade camber angle $\alpha_2 = -61.7°$. This statement is verified by considering the dimensionless pressure integral, which reflects the blade lift coefficient. Taking the solidity and the inlet flow angle constant, the measured blade loading is achieved by increasing the exit flow angle. The cascade calculations using the given $\alpha_2$ produced less than the measured blade loading. Therefore calculations were made to find a deviation angle $\delta$ to use in the cascade calculations so that they would match the experimental measurements. It was found that a deviation of $\delta = -2.5°$, which gives in an exit angle $\alpha_2 = -64.2°$, provides the best agreement between calculation and measurement. Figure 3 shows the calculation results, which are in a good agreement with the experimental results. Pressure distributions were calculated for Reynolds numbers $Re = 3.65\times10^6$ and $1.33\times10^6$ assuming that the above deviation will not significantly vary with the Reynolds number. Using the above angle, the pressure distribution for the Reynolds number $Re = 3.6\times10$ was plotted in Fig. 4 that show good agreement with experimental data. Similar good results are obtained for Reynolds number $Re = 1.33\times10^6$. This good agreement created confidence in the pressure calculation method that generates reliable velocity distribution necessary for heat transfer calculations.

Viscous Flow: Heat Transfer, Boundary Layer Analysis

The heat transfer experiment was analyzed for three different Reynolds number listed in Table 2. The corresponding calculated pressure distributions like those in Figs. 3 and 4 were implemented into the heat transfer calculation procedure. In accordance with the experimental data reported by Yeh et al. (1990), a free stream turbulence level $Tu = 0.021$ was used. To account for the turbulent character of the flow in the calculation procedure, the differential code STAN5 (Crawford and Kays, 1976) serves as the calculation tool. A discussion of the experimental and calculated heat transfer results, and the boundary layer calculation follow.

Heat transfer coefficient. All calculations were made using the STAN5 boundary layer calculation. Only the turbulence and transition model available in STAN5 were used. The implementation and form of these models has been described by Gaugler (1981).

The first case investigated has the lowest Reynolds number. Starting with $Re = 1.3\times10^6$, the corresponding inlet flow conditions was taken from Table 2, the pressure distribution, and the turbulence intensity of $Tu = 0.021$ was used. As previously indicated, because of the lack of appropriate transition models the apriori prediction of heat transfer coefficients is hardly possible. Thus a number of iterations were necessary to match the experimental and calculation results. In order to define the iteration range, two calculations are shown in Fig. 5 as curves 1 and 2. Curve 1, incorporates the Dunham (1972) transition start and the Dhawan and Narasimha (1958) transition length correlation, curve 2, utilizes the abrupt transition. The calculated heat transfer data for the suction surface (SS) are enclosed by curves 1 and 2 representing the above transition models. As shown, the results of calculation for both curves significantly differ from the experimental data. To define a reasonable transition start, the average between the location of the maximum on curve 2 and the minimum on curve 1 is taken and

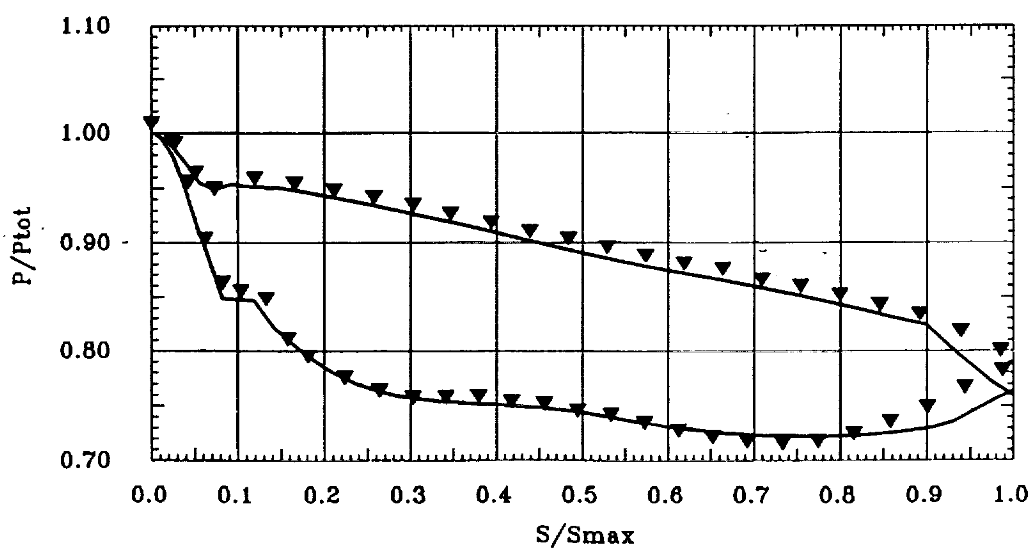


FIGURE 3. - MEASURED (▼) AND CALCULATED RELATIVE PRESSURE AS A FUNCTION OF RELATIVE SURFACE DISTANCE FOR THE EXIT CAMBER ANGLE $\alpha_2 = -64.2^0$ AND THE REYNOLDS NUMBER $Re = 6.3\times10^6$.

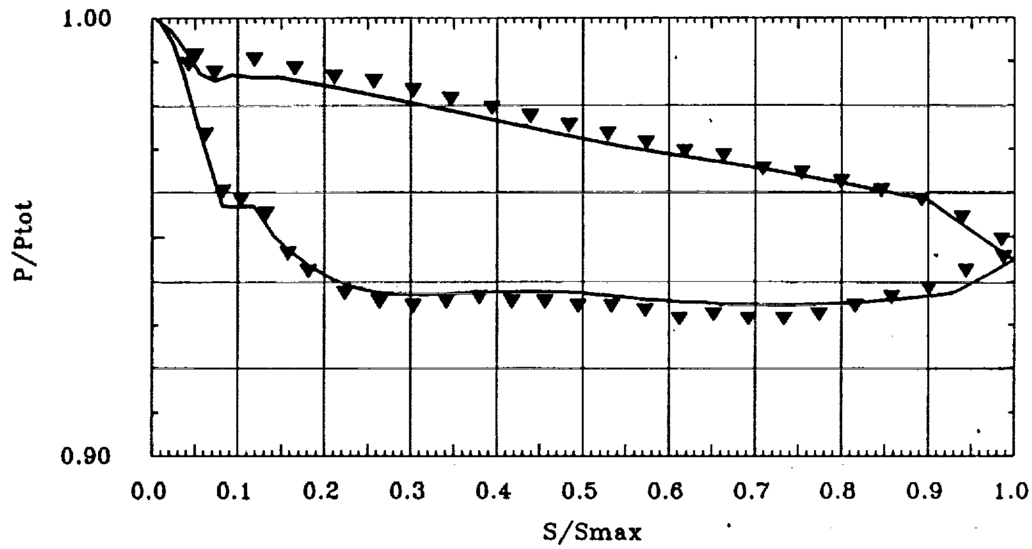


FIGURE 4. - MEASURED (▼) AND CALCULATED RELATIVE PRESSURE AS A FUNCTION OF RELATIVE SURFACE DISTANCE FOR THE EXIT CAMBER ANGLE $\alpha_2$ = -64.2° AND THE REYNOLDS NUMBER Re = $6.3\times10^6$.

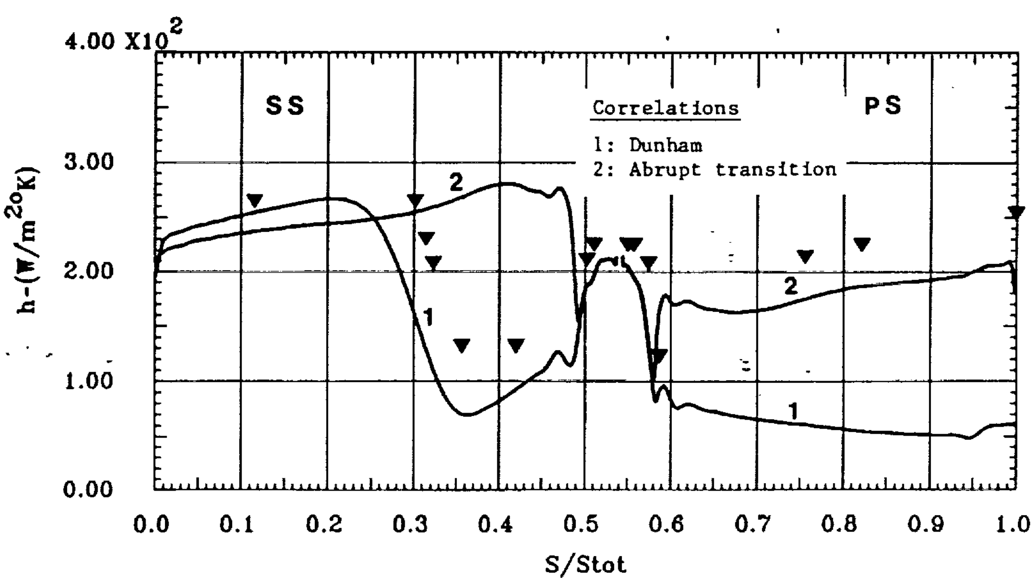


FIGURE 5. - MEASURED (▼) AND CALCULATED HEAT TRANSFER COEFFICIENT AS A FUNCTION OF RELATIVE SURFACE DISTANCE. Re = $1.3\times10^6$.

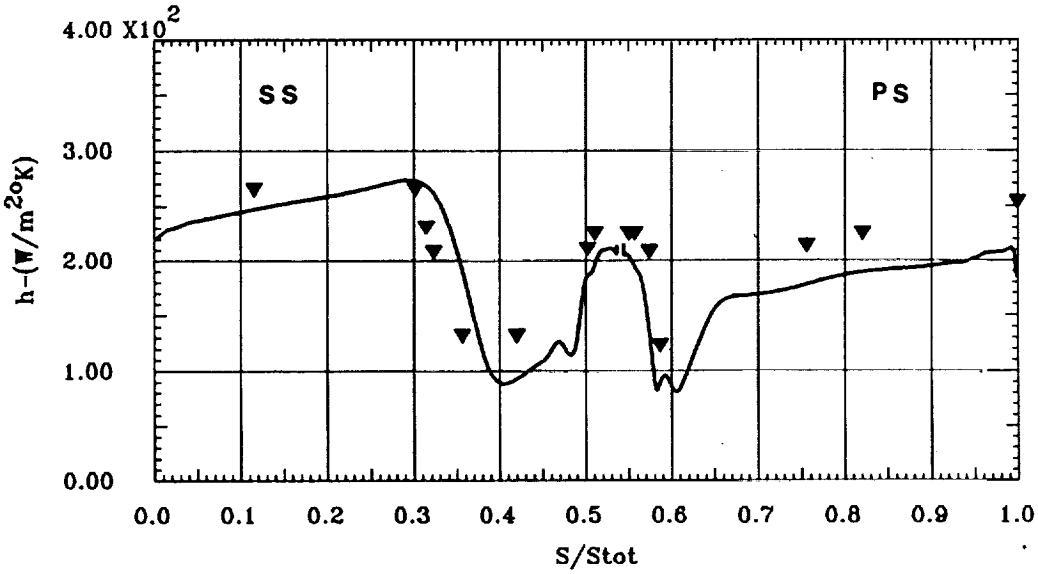


FIGURE 6. - MEASURED (▼) AND CALCULATED HEAT TRANSFER COEFFICIENT AS A FUNCTION OF RELATIVE SURFACE DISTANCE. FORCED TRANSITION, START ON SUCTION AND PRESSURE SURFACE AT S/STOT = 0.433 AND 0.54, FOR REYNOLDS NUMBER Re = $1.3\times10^{6}$.

implemented into the next calculation. On the pressure surface (PS), the calculated heat transfer coefficient, curve 1, indicates no transition over the entire surface. This circumstance results in a considerably lower heat transfer coefficient than measured. Assuming an abrupt transition, curve 2, the calculation shows a significant shift towards the experimental data with a remaining difference of approximately 20 percent over the entire pressure surface. This difference is probably attributed to the oversimplification of energy equation. Attempts to reduce the discrepancy between calculation and measurement by using the above mentioned averaging, the forced transition start at the relative distance of S/Stot = 0.433 has led to significantly better results on the suction surface as shown in Fig. 6. On the pressure surface a shift of transition start to S/Stot = 0.54 has brought only a marginal improvement.

The second case investigated pertains to an intermediate Reynolds number Re = $3.67\times10^{6}$. Reasonable results were obtained for this Reynolds number. The turbulence intensity was the same as the first case. As Fig. 7 shows, the heat transfer coefficient is calculated using alternatively Abu-Ghannam and Shaw (1980), VanDriest and Blumer (1963), and Dunham start correlations by keeping the Dhawan and Narasimha's transition length model. For this particular case and only for suction surface (SS), the VanDriest and Blumer model seems to generate results that are slightly closer to the experimental data. On the pressure surface, however, the application of the above models lead to almost identical results that significantly differ from experimental data. Using the VanDriest and Blumer start correlation as an appropriate model for this particular case, the heat transfer coefficient is calculated using alternatively Chen and Thyson (1971), Dhawan and Narasimha, and the abrupt transition models. As shown in Fig. 8, on the suction surface, the correlation of Chen and Thyson, curve 1, is slightly closer to the experimental data. On the pressure surface, the abrupt transition, curve 3, apparently describes the situation more realistically. The results of the above iterative process in coincidence with those by Gaugler (1985) show clearly that none of the correlations discussed is capable of predicting the transition mechanism. Consequently none of those can be recommended for predicting the heat transfer coefficient.

The third case utilizes the highest Reynolds number Re = $6.36\times10^{6}$. Similar to the cases treated previously, for this case, the corresponding inlet flow conditions from Table 2, the pressure distribution displayed in Fig. 3, and the turbulence intensity Tu = 0.021 were used. As mentioned previously, the heat transfer data at this Reynolds number is considered questionable. For the analysis of this case, the Dunham transition-start and the Dhawan and Narasimha length correlations were used. As shown in Fig. 9, there is a substantial disagreement between the experimental and calculation results. The uncertainties in measured data, the simplification of the governing equations, the deficiencies in accurately predicting the transition location, and length have contributed to this extreme discrepancy in this case.

Boundary layer results. The experimental investigations were restricted to pressure and heat transfer measurements. However, in connection with this analysis, a brief discussion of boundary layer calculation results is necessary to explain some discrepancies encountered in heat transfer analysis discussed above. Accurate prediction of boundary layer development along the blade surface is the first condition for a reliable

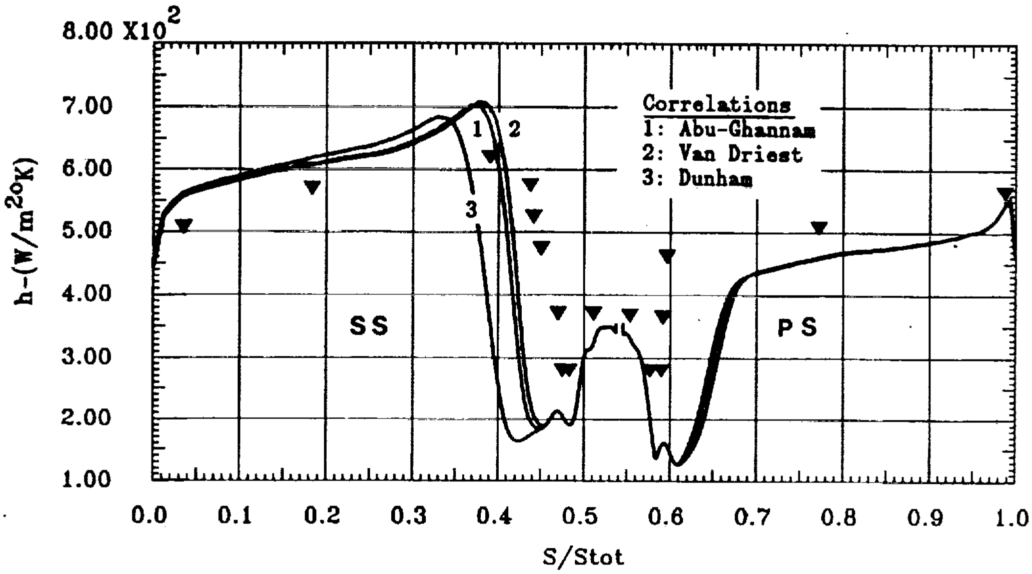


FIGURE 7. - MEASURED (▼) AND CALCULATED HEAT TRANSFER COEFFICIENT AS A FUNCTION OF RELATIVE SURFACE DISTANCE USING THREE DIFFERENT TRANSITION START MODELS. LENGTH CORRELATION FROM DHAWAN AND NARASIMHA. $Re = 3.7\times10^6$.

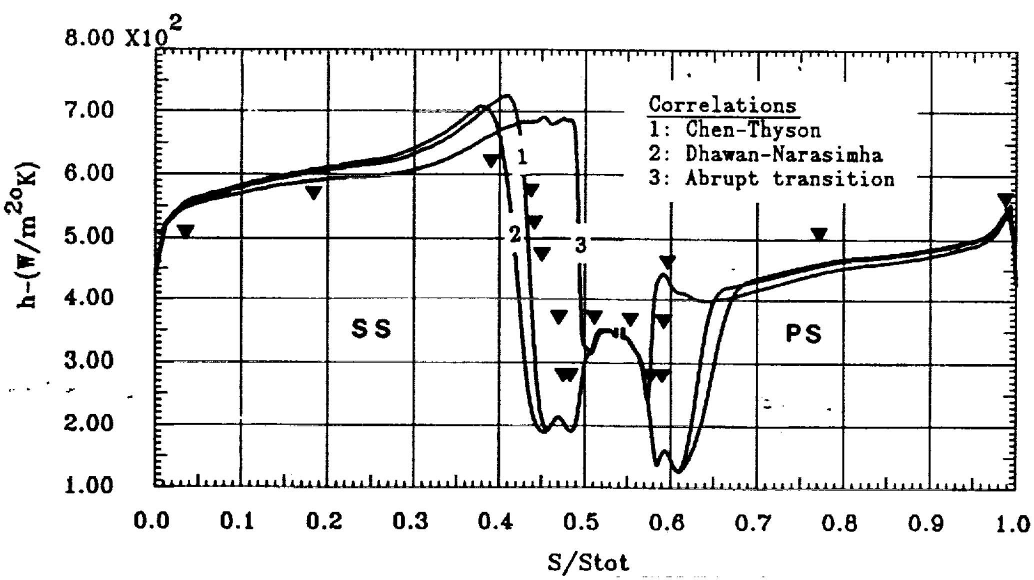


FIGURE 8. - MEASURED (▼) AND CALCULATED HEAT TRANSFER COEFFICIENT AS A FUNCTION OF RELATIVE SURFACE DISTANCE USING DIFFERENT TRANSITION LENGTH MODELS. START CORRELATION FROM VAN DRIEST AND BLUMER. $Re = 3.7\times10^6$.

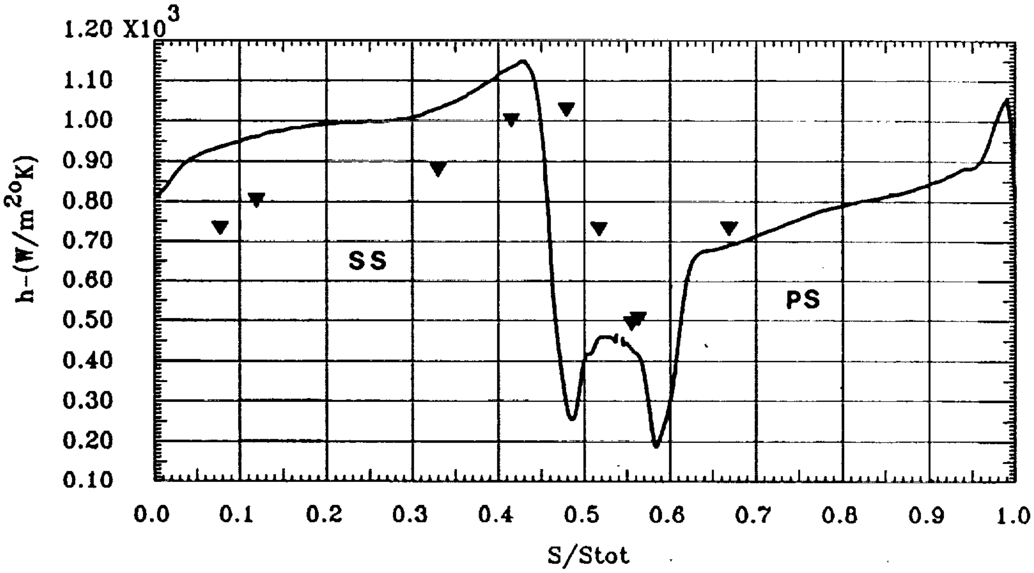


FIGURE 9. - MEASURED (▼) AND CALCULATED HEAT TRANSFER COEFFICIENT AS A FUNCTION OF RELATIVE SURFACE DISTANCE. TRANSITION START CORRELATION FROM DUNHAM. TRANSITION LENGTH CORRELATION FROM DHAWAN AND NARASIMHA. Re = $6.3\times10^6$.

prediction of the heat transfer and friction coefficients. The existing differential boundary layer and heat transfer calculation methods incorporate continuity, momentum, and energy differential equations together with turbulence models. For an incompressible flow with small temperature changes (negligible influence of temperature on viscosity), the first two equations are decoupled from the energy equation. The energy equation, however, is strongly linked with these two equation. Consequently, any changes in velocity distribution affect the temperature and therefore the heat transfer calculation. By looking at the heat transfer coefficients (Figs. 5 to 9), one encounters several low amplitude oscillations near the leading edge on the suction and pressure surface. The location of these oscillations exactly correspond to those occurring in the boundary layer calculation. The effect of abrupt changes of velocity distributions are indirectly seen in Figs. 10 and 11 which display the course of displacement and momentum thicknesses for Re = $3.7\times10^6$. As shown in Fig. 11, on the suction surface at S/Stot ≈ 0.5, the displacement thickness experiences the first steep increase resulting in a significant underprediction of heat transfer coefficient that extends from S/Stot = 0.5 to 0.485 in Fig. 7. The overprediction at S/Stot ≈ 0.38 in Fig. 7 is triggered by the oscillation at S/Stot ≈ 0.48 shown in Fig. 11. These oscillations are produced numerically and have no physical foundation.

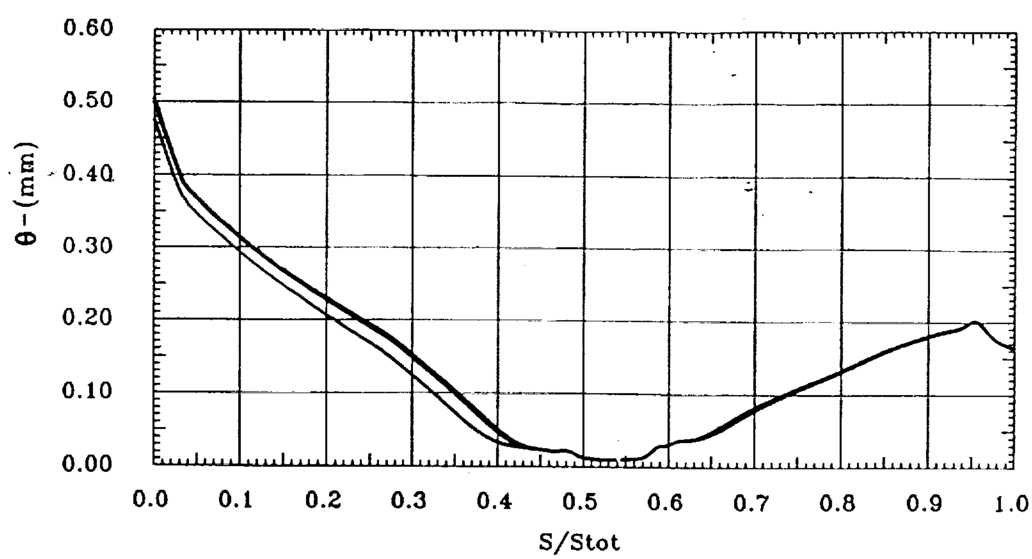


FIGURE 10. - BOUNDARY LAYER MOMENTUM THICKNESS AS A FUNCTION OF RELATIVE SURFACE DISTANCE USING THREE DIFFERENT TRANSITION START MODELS. LENGTH CORRELATION FROM DHAWAN AND NARASIMHA. Re = $3.7\times10^6$.

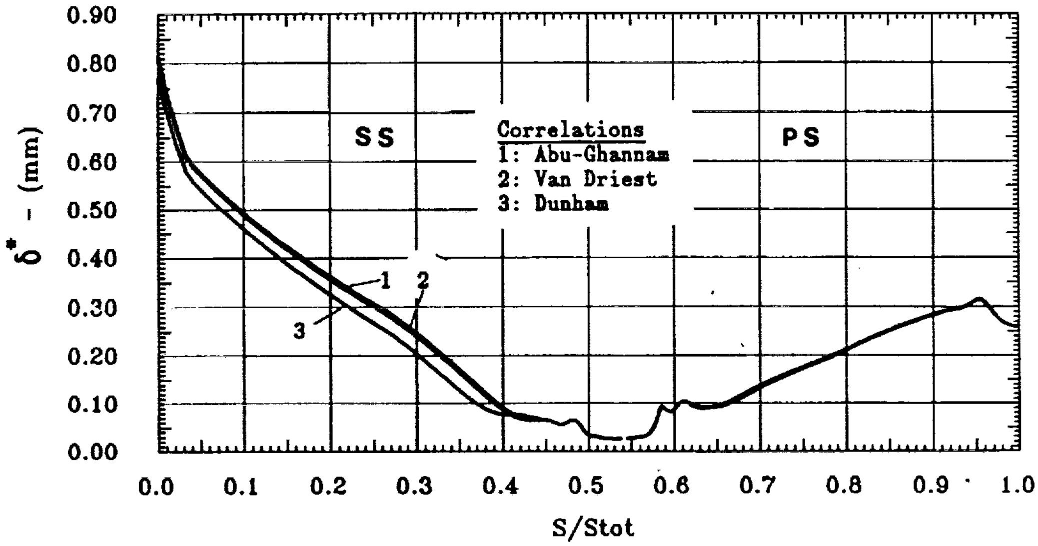


FIGURE 11. - BOUNDARY LAYER DISPLACEMENT THICKNESS AS A FUNCTION OF RELATIVE SURFACE DISTANCE USING THREE DIFFERENT TRANSITION START MODELS. LENGTH CORRELATION FROM DHAWAN AND NARASIMHA. $Re = 3.7 \times 10^6$.

## CONCLUSIONS

The conclusions from this investigation concern (1) test facility modifications, (2) measurement technique, and (3) calculation method.

1. It was shown that the pressure distribution within a cascade can be simulated reasonably well by using a single blade test section, provided that the exit flow angle of the test section coincides with that of a cascade. At the exit station 2, the existing test section has a sudden expansion that has affected exit flow angle required for a periodic cascade flow. This sudden expansion should be eliminated as shown in Fig. 12. Also at station 2, the concave channel wall constricts the flow in what is normally the uncovered portion of a cascade flow path. The concave channel wall downstream of this station 2 needs to be adjustable if cascade blade loadings are to be accurately simulated.

The convex and concave walls with leading edges were initially thought to simulate the leading edges of a cascade. However, they do not serve this purpose for the reason explained in the text. To obtain a well defined inlet velocity profile, these edges should be eliminated as shown in Fig. 12.

For theoretical analysis and validation, it is helpful to have the information about the inlet and exit flow and velocity distributions. This requires a traversing measurement of the flow conditions including flow angle at the inlet and exit.

2. This study has shown that the liquid crystal temperature measurement technique is a simple and reliable tool and should be further developed. This technique enables measuring the temperature at any arbitrary point such as transition point. Automation of data acquisition and processing is highly recommended.

3. The analysis has shown that the code STAN5 is not capable of apriori predicting the heat transfer coefficient reliably and quantitatively. Only qualitative results were obtained by performing several

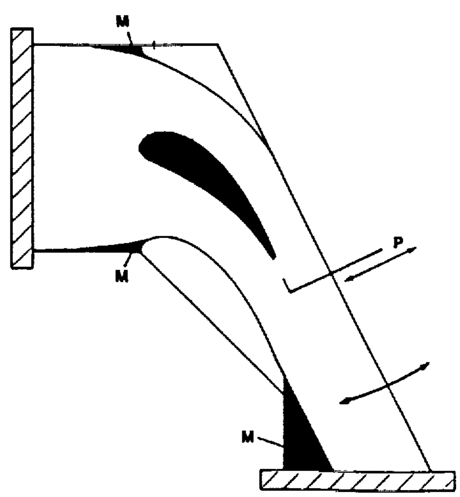


FIGURE 12. - MODIFIED TEST SECTION. M MARKS THE MODIFICATIONS. P = PROBE.

iterations and adjustments. The key weakness of this code is the lack of an appropriate transition model. This deficiency together with the oversimplification of energy equation have contributed to over/underprediction of heat transfer coefficient. The results underscore the urgent need for establishing a comprehensive research program on transition phenomenon. The existing empirical models may be capable of representing some special cases, they do not describe the transition phenomenon.

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## REFERENCES

Abdul-Aziz, A., Tong, M.T., and Kaufman, A., 1989, "Thermal Finite-Element Analysis of Space Shuttle Main Engine Turbine Blade," Finite Elements in Analysis and Design, Vol. 5, No. 4, pp. 337-348.

Abu-Ghannam, B.J., and Shaw, R., 1980, "Natural Transition of Boundary Layers, the Effect of Turbulence, Pressure Gradient, and Flow History," Journal of Mechanical Engineering Science, Vol. 22, No. 5, pp. 213-228.

Chen, K.K., and Thyson, N.A., 1971, "Extension of Emmons' Spot Theory to Flows on Blunt Bodies," AIAA Journal, Vol. 9, No. 5, pp. 821-825.

Dunham, J., 1972, "Predictions of Boundary Layer Transition on Turbomachinery Blades," Boundary Layer Effects in Turbomachines, J. Surugue, ed., AGARD, Paris, France, AGARD-AG-164, pp. 55-71.

Dhawan, S., and Narasimha, R., 1958, "Some Properties of Boundary Layer Flow During the Transition from Laminar to Turbulent Motion," Journal of Fluid Mechanics, Vol. 3, pp. 418-436.

Gaugler, R.E., 1981, "Some Modifications to, and Operational Experiments with, the Two-Dimensional, Finite-Difference, Boundary-Layer Code, STAN5," NASA TM-81631.

Gaugler, R.E., 1985, "A Review and Analysis of Boundary Layer Transition Data for Turbine Application," ASME Paper 85-GT-83 (also, NASA TM-86880).

Hippensteele, S., and Russell, L.M., 1988 "High-Resolution Liquid-Crystal Heat Transfer Measurements on the End Wall of a Turbine Passage with Variations in Reynolds Number," NASA TM-100827.

Katsanis, T., and McNally, W.D., 1977, "Revised FORTRAN Program for Calculating Velocities and Streamlines on the Hub-Shroud Mid-channel Stream Surface on an Axial-, Radial-, or Mixed-Flow Turbomachine or Annular Duct, I-User's Manual," NASA TN D-8430.

McFarland, E.R., 1985, "A FORTRAN Computer Code for Calculating Flows in Multiple-Blade-Element Cascades," NASA TM-87104.

Patankar, S.V., and Spalding, D.B., 1970, Heat and Mass Transfer in Boundary Layers, 2nd Edition, Intertext Books, London.

Runchal, A.K., 1974, "Blade With Transpiration, Film Cooling and Transition - The BTFT Computer Program," CHAM Technical Report 880/1, Combustion Heat and Mass Transfer Ltd., Wimbleton, England.

Spalding, D.B., 1977, GENMIX: A General Computer Program for Two-Dimensional Parabolic Phenomena, Pergamon Press.

Spalding, D.B., 1973, "A General Computer Program for Two-Dimensional Boundary Layer Problems," Report HTS/73/48, Imperial College of Science and Technology, Department of Mechanical Engineering, London, England.

VanDriest, E.R., and Blumer, C.B., 1963, "Boundary Layer Transition: Free Stream Turbulence and Pressure Gradient Effects," AIAA Journal, Vol. 1, pp. 1303-1306.

Vavra, M.H., 1960, Aero-Thermodynamics and Flow in Turbomachines, John Wiley and Sons, Inc., New York.

Yeh, F., Lopez, I., and Hippensteele, S., 1990, "High Reynolds Number Heat Transfer Experiment," presented at the Fourth Conference on Advanced Earth-To-Orbit Propulsion Technology, Marshal Space Flight Center, Huntsville, AL (to be published as a NASA Conference Publication).

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16. Abstract

In this report the results of aerodynamic and heat transfer experimental investigations performed in a high Reynolds number turbine cascade test facility are analyzed. The experimental facility simulates the high Reynolds number flow conditions similar to those encountered in the space shuttle main engine. In order to determine the influence of Reynolds number on aerodynamic and thermal behavior of the blades, heat transfer coefficients were measured at various Reynolds numbers using liquid crystal temperature measurement technique. Potential flow calculation methods were used to predict the cascade pressure distributions. Boundary layer and heat transfer calculation methods were used with these pressure distributions to verify the experimental results.

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